

Mechanical discrete simulator of the electro-mechanical lift with n:1 roping

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Abstract. The design process of new products in lift engineering is a difficult task due to, mainly, the complexity and slenderness of the lift system, demanding a predictive tool for the lift mechanics. A mechanical ad-hoc discrete simulator, as an alternative to ‘general purpose’ mechanical simulators is proposed. Firstly, the synthesis and experimentation process that has led to establish a suitable model capable of simulating accurately the response of the electromechanical lift is discussed. Then, the equations of motion are derived. The model comprises a discrete system of 5 vertically displaceable masses (car, counterweight, car frame, passengers/loads and lift drive), an inertial mass of the assembly tension pulley-rotor shaft which can rotate about the machine axis and 6 mechanical connectors with 1:1 suspension layout. The model is extended to any n:1 roping lift by setting 6 equivalent mechanical components (suspension systems for car and counterweight, lift drive silent blocks, tension pulley-lift drive stator and passengers/load equivalent spring-damper) by inductive inference from 1:1 and generalized 2:1 roping system. The application to simulate real elevator systems is proposed by numeric time integration of the governing equations using the Kutta-Meden algorithm and implemented in a computer program for ad-hoc elevator simulation called ElevaCAD.

1. Introduction

The analysis of the dynamic behavior of the elevator car system plays an important role in elevator engineering and superior ride quality of elevators is demanded nowadays. In particular, vibration in the low frequency range must be investigated in the modern design of elevator systems. International standard encourage the development of uniform, reliable and precise measurement and processing techniques to be applied within the elevator industry [1]. It is a challenging task to develop a precise and reliable tool/software in order to simulate the dynamic characteristics of the elevator system. In addition to research, education and training applications could be improved with a tool like that.

The use of general purpose software packages [2-4] is the unique alternative when elevator engineers need to simulate the dynamics of the elevator, at present, if an elevator test tower or any other facility for experimentation is not available. The implementation of a mechanical model in a software package entails the modeling of the system components. The basic assumptions on the elevator system dynamics are postulated in order to get a precise response in standard elevator trip and residential building elevators in a first stage. So the previous work [5] was devoted to study the elevator dynamics by testing in real installations and take conclusions on the trends of the system response and critical components parameters. It was detected that fine variations of those parameters



greatly influenced the overall response, so a list of such parameters and specific test procedures were developed and done [6].

The main and critical components of the elevator system are the passengers. As it is well known, the behavior of passengers is very complicated factor to accommodate in the elevator model [7-9]. Further on, the passengers are the most important “mechanical components” of the elevator system. The task for modeling the passengers could be even more challenging than that of the elevator system itself. Recently, elevator models and measurement procedures have been researched [10-11] to determine the damping and stiffness characteristics of passengers.

Dedicated specialist multibody dynamics software packages like Adams [2], SolidWorks [3] and Working Model [4] can be applied to study the dynamic response of the elevator. Implicitly, the use of simulators means idealization so some mechanic model of the elevator must be defined. This paper deals with the process and justification to derive an optimized valid mechanical model of the elevator system as a whole, based on rope dynamics analysis and testing results on in-situ elevators.

2. The elevator system

In a typical elevator system the car frame is suspended by sling and the car is mounted within the car frame on isolation blocks (see figure 1). The elevator is inside a building hole known as shaft and supported by a number of solutions to the building structure. In figure 1 the lift drive, the diverter pulley and its bed is isolated from the building wrought by silent blocks. Minimum torque requirement makes quite common to suspend also a counterweight by the other end of the sling.

The car frame and the counterweight are forced by the guide rails to move substantially in the vertical direction, without twist, by roller guides or sliding guide shoes components. In the elevator system, vibrations in both lateral and vertical directions are of interest. Lateral and vertical displacements of the car are important for comfort and noise control but vertical displacements are essential. Vertical vibrations are determinant to the dynamic response, safety, durability, fatigue, maintenance and comfort of the elevator system.

The car, car-frame, and counterweight are generally made of metallic pieces and fixed to each counter-part by bolts and nuts or welding techniques. Special attention is devoted to tighten the pieces to avoid noise during travel. Every of these components almost constitute an ideal rigid solid. The six degrees of freedom of a rigid solid are reduced to only one, thus assuming ideal sliding junctions between car-frame or counterweight and the guide rails.

The components of the elevator system has been studied separately and derivation of a simplified one degree of freedom model was supported by experiments.

The response of the suspension ropes is essential to understand and model the elevator system. The sling is generally made by a number n_R of tension equalized metallic ropes disposed in series, fixed to the car-frame by one end and to the counterweight by the other end. The contact between the rope and traction sheave and diverter pulleys is specially critical, not only for the necessity to produce the desired movement of the car but its control, safety, and adherence response too.

3. The proposed model

According to figure 2 we assign the subscript (*CW*) to the counterweight, (*FR*) to the car frame, (*CA*) to the car, (*PA*) to the weights/passengers travelled, and (*MA*) to the drive including its foundation. They all are considered rigid bodies. Each of these bodies is assigned one vertical degree of freedom y_{CW} , y_{FR} , y_{CA} , y_{PA} and y_{MA} (See figure 2).

Let's assign the variable y_{FR} for the vertical displacement of the car-frame and the variable y_{CW} for that of the counterweight. The car is fixed to the car-frame by four silent-blocks. This makes the car to be movable in the vertical direction. The variable y_{CA} is assigned. We also must consider the most important part of the elevator that the passengers or goods are. We assign for simplicity variable y_{PA} to the vertical displacement of the elevator continent.

The structure of the machine drive is composed of two rigid parts the drive stator-drive foundation and the shaft which integrates the traction sheave, rotor, brakes (BR) and the flywheel. The shaft can rotate almost ideally. So we conceive the shaft rotation α as one degree of freedom of the elevator system. The machine drive-foundation substructure is isolated from its base to avoid noise transmission to the building structure then is assigned an extra degree of freedom to its vertical displacement y_{MA} . The product of the shaft rotation α by the radius of the traction sheave R is the displacement of a peripheral point of the traction sheave αR which is used instead of α to get dimensional homogeneity of the displacement vector $[y]$. The drive brakes (BR) when actuated are contemplated in the model by an equivalent spring damper (BR) of stiffness k_{BR} and damping c_{BR} .

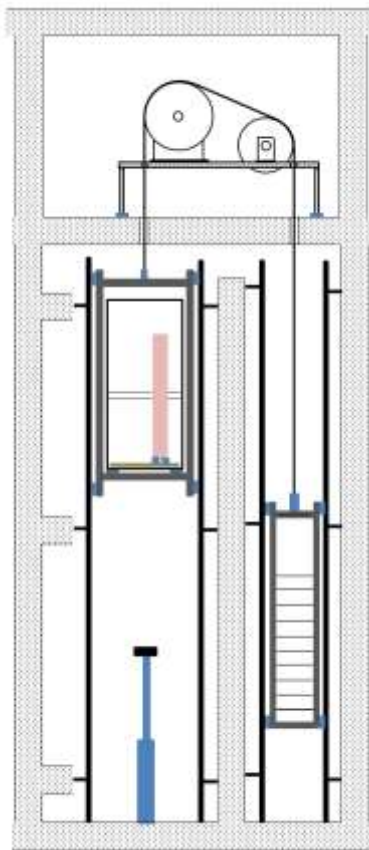


Figure 1. Elevator system scheme.

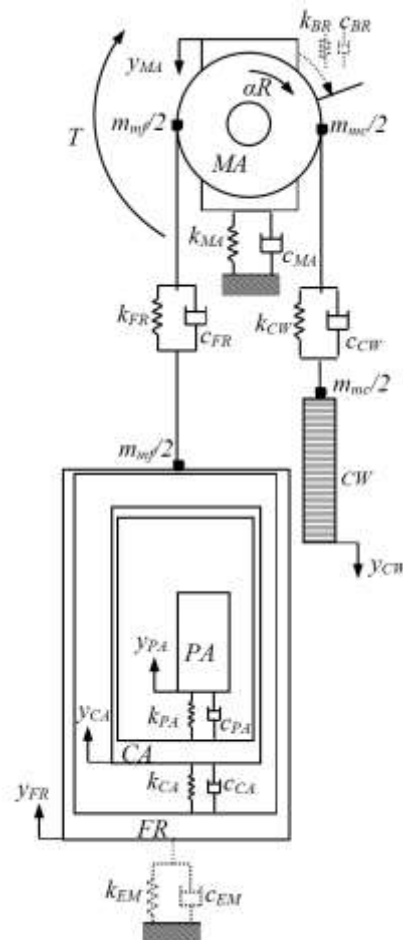


Figure 2. 6 d.o.f mechanical model of the elevator car system.

Simulation of the impact of the car-frame against the buffers or safety gear actuation is suggested by an equivalent spring/damper (EM) of stiffness k_{EM} and damping c_{EM} .

If the car is joined by several silent-blocks (or other type of similar low stiffness dampers) to the car frame, an equivalent system where the car and the car frame are connected by an equivalent spring/damper of stiffness k_{CA} and damping c_{CA} is suggested. If we consider that the car of mass m_{CA} is attached to the car frame by a number of isolation blocks located at each corner of the car floor (parallel configuration of the individual silent-blocks) the resulting stiffness and damping coefficients for the car are:

$$k_{CA} = n_{ib} k_{ib}; c_{CA} = n_{ib} c_{ib} \quad (1)$$

where n_{ib} is the number of isolation blocks used in the car and k_{ib} and c_{ib} are the stiffness and viscous damping coefficients of a single isolation block.

It is conventionally assumed undamped condition in many analysis with ropes. The available decay data of damping tests on ropes [12] and specially designed tests previously researched [10-11] have revealed that rope undamping is not acceptable for the elevator's mechanical modelling we are driving. The suspension ropes possess some rigidity to bend when subjected to loads of the order of 1/12 times the one for reaching its elastic limit (12 is a typical safety factor for in-service three ropes suspensions for attaining fatigue criteria that EN-81-20 normalizes). That means that the composition of the rope influences its dynamics performance by means of a non constant damping factor ζ which is affected to some extent by the suspended load.

The rope-pieces connecting the traction pulley and the car frame of mass m_{FR} is equivalent to a linear spring/damper of stiffness k_{FR} and damping c_{FR} , length l_f and a number of parallel suspension ropes n_R :

$$k_{FR} = n_R \frac{EA}{l_f(t)}; c_{FR} = 2\zeta \sqrt{\frac{n_R EA}{l_f(t)} (m_{FR} + m_{CA} + m_{PA})} \quad (2)$$

where E is the apparent elastic modulus of a single suspension rope, A is its true sectional area, and ζ is the damping factor of a single rope which it is assumed to be tensioned by a suspended mass of $(m_{FR} + m_{CA} + m_{PA})/n_R$. The damping factor can be computed from a series of tests with a single rope of the same type of any length and suspended mass and assumed constant.

Similarly, the rope-piece connecting the traction pulley and the counterweight of mass m_{CW} is equivalent to a linear spring/damper of constants:

$$k_{CW} = n_R \frac{EA}{l_c(t)}; c_{CW} = 2\zeta \sqrt{\frac{n_R EA}{l_c(t)} m_{CW}} \quad (3)$$

where l_c is the length of the sling rope-piece, and ζ is the damping factor of a single rope when subjected to tension by a suspended mass of m_{CW}/n_R .

The rope mass is also studied. The rope-piece mass connecting the frame to the traction sheave is denoted by m_{mf} , and the rope-piece mass connecting the counterweight to the traction sheave is denoted by m_{mc} . These two masses can be easily determined as a function of the rope density ρ , effective cross section A , the number of suspension ropes n_R and its instantaneous lengths $l_f(t)$ and $l_c(t)$, as follows:

$$m_{mf} = n_R \rho A l_f(t); m_{mc} = n_R \rho A l_c(t) \quad (4)$$

The simplified model joins half the mass of the sling frame to the car-frame mass. Then, the remaining half is joint to the periphery of the traction drive. The same mass distribution applies to the counterweight sling. In this way, the computed static deformations of the elevator components in the reference configuration of the system are in agreement with Strength of Materials formulas [14] without increasing model complexity [13].

Also, the drive-foundation silent blocks defines mainly the equivalent joint between the machine stator and the building structure/ground k_{MA} and c_{MA} assuming that if several massless springs joints are disposed in series, the equivalent spring can be approximated by the lowest individual stiffness.

The passengers are joined to the car mass, m_{CA} , through two stiffless mass components: the shoes (sh) and the covering car floor (fl). Both can be treated separately as ideal spring/dumper joints. Then, it is assumed [7,8] that the human body spring/damper in the vertical direction, the shoes and the floor spring/dampers are connected in series, and can be reduced to one ideal spring/damper joint whose constants are k_{lp} and c_{lp} , respectively.

$$\frac{1}{k_{lp}} = \frac{1}{k_{pa}} + \frac{1}{k_{sh}} + \frac{1}{k_{fl}}; \frac{1}{c_{lp}} = \frac{1}{c_{pa}} + \frac{1}{c_{sh}} + \frac{1}{c_{fl}} \quad (5)$$

where k_{lp} denotes the overall equivalent stiffness coefficient of one passenger in the vertical direction, k_{pa} the equivalent stiffness of one passenger in bear feet position, k_{sh} the shoes-stockings vertical stiffness coefficient of one passenger, and k_{fl} the vertical stiffness coefficient of the car floor. Similarly, c_{lp} , c_{pa} , c_{sh} and c_{fl} denote the overall equivalent vertical viscous damping for one passenger, the equivalent vertical viscous damping coefficients for the passengers, the shoes-stockings coefficients and the car floor coefficient.

Assuming that all the passengers travelling in the car move in phase relative to each other and have the same mass with the average passenger's mass being m_{lp} , the resulting coefficients k_{PA} and c_{PA} for all passengers in the car are:

$$k_{PA} = n_p k_{lp}; c_{PA} = n_p c_{lp}; m_{PA} = n_p m_{lp} \quad (6)$$

where n_p is the number of passengers travelling in the car. Similar rules apply if goods alone or mixed passengers/goods are carried by the car.

4. n:1 roping equations

Following 2nd Newton's law the governing equation of the system of rigid solids shown in figure 2 can be written and generalized to n:1 roping:

$$[M][\ddot{y}] + [C][\dot{y}] + [K][y] = [F] \quad (7)$$

where $[M]$ is the mass matrix, $[C]$ is the damping matrix, $[K]$ is the stiffness matrix, $[F]$ is the time dependent vector of applied forces and $[y]$ is the time displacement vector. The mass matrix has the following form:

$$[M] = \begin{bmatrix} \frac{I_{RM}}{R^2} + \frac{m_{mf}}{3} + \frac{m_{mc}}{3} + \sum_{d=1}^{n_d} \frac{I_d}{r_d^2} & \frac{m_{mf}}{6} & 0 & 0 & \frac{m_{mc}}{6} & 0 \\ \frac{m_{mf}}{6} & m_{FR} + \frac{m_{mf}}{3} & 0 & 0 & 0 & 0 \\ 0 & 0 & m_{CA} & 0 & 0 & 0 \\ 0 & 0 & 0 & m_{PA} & 0 & 0 \\ \frac{m_{mc}}{6} & 0 & 0 & 0 & m_{CW} + \frac{m_{mc}}{3} & 0 \\ 0 & 0 & 0 & 0 & 0 & m_{MA} + \frac{m_{mf}}{2} + \frac{m_{mc}}{2} \end{bmatrix}; [y] = \begin{bmatrix} \alpha R \\ y_{FR} \\ y_{CA} \\ y_{PA} \\ y_{CW} \\ y_{MA} \end{bmatrix} \quad (8)$$

where I_{RM} is the shaft inertia which integrates the traction sheave, brakes, rotor and flywheel, R is the traction sheave radius, n_d is the number of diverter pulleys, I_d the inertia of the d -th diverter pulley, r_d is the radius of the d -th diverter pulley, m_{MA} is the machine mass including the traction sheave mass, m_{FR} is the car-frame mass, m_{CA} is the car mass, and m_{PA} is the mass of the passengers or goods.

The distributed mass that the slings rope-pieces are, has been included in the model by assembling the governing equations of the consistent mass approach [13]. This apply two both the rope-piece connecting the frame to the drive which mass is denoted by m_{mf} , and the rope-piece connecting the counterweight to the drive which mass is denoted by m_{mc} . These two masses were previously derived as a function of its instantaneous lengths $l_f(t)$ and $l_c(t)$ as shown in equation (4). The instantaneous lengths can be computed as a function of the roping ratio n , the instantaneous car frame position $y_{CA}(t)$, the maximum trip distance from 0-floor to n -th floor h , and the minimum free installation lengths of the frame rope-piece s_{of} (car stopped in the n -th floor) and the counterweight rope-piece s_{oc} (for the car stopped in the 0 floor) by:

$$\begin{aligned} l_f(t) &= s_{0f} + n(h - y_{CA}(t)) \\ l_c(t) &= s_{0c} + ny_{CA}(t) \end{aligned} \quad (9)$$

The minimum free installation lengths s_{0f} and s_{0c} can be estimated from h_f and h_w shown in the installation layout (see figure 3) by $s_{0f} \approx nh_f$ and $s_{0c} \approx nh_w$. It was considered that the small pieces of rope of the slings in contact with the traction sheave and diverter pulleys may not be included in the distributed mass of the slings m_{mf} and m_{mc} , nor influence its stiffness k_{fr} and k_{cw} and damping c_{fr} and c_{cw} . See how the linear rope speed in the counterweight sling $\dot{l}_c(t)$ is positive and how, in the frame sling $\dot{l}_f(t)$, is negative in an ascending trip ($\dot{y}_{CA}(t) \geq 0$):

$$n\dot{y}_{CA}(t) = -\dot{l}_f(t) = \dot{l}_c(t) \quad (10)$$

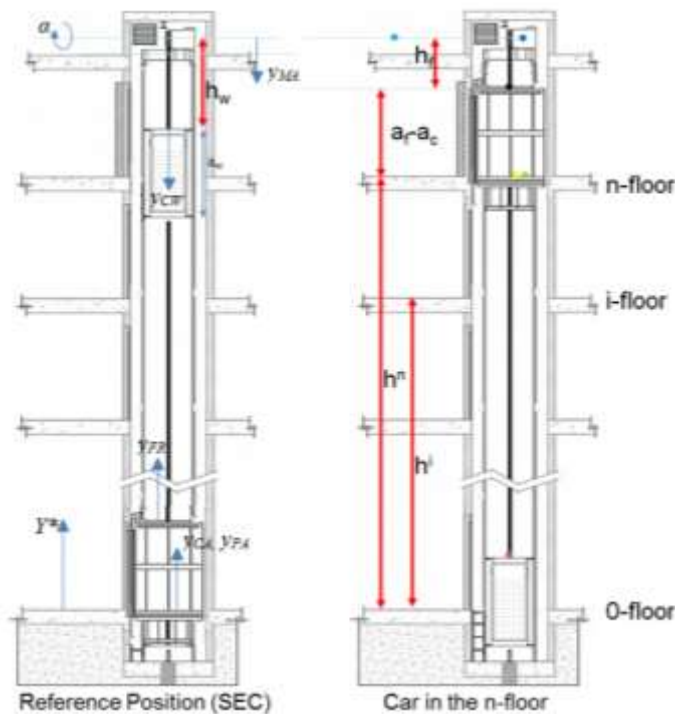


Figure 3. Installation parameters and origin of displacement of the system proposed.

and how its absolute value is n times greater according to the $n:1$ roping of the elevator if conventional differential no resistance to rotation pulleys are used.

The stiffness matrix $[K]$ of the elevator system has the form:

$$[K] = \begin{bmatrix} k_{CW} + k_{FR} + k_{BR} & -k_{FR} & 0 & 0 & -k_{CW} & k_{CW} - k_{FR} \\ -k_{FR} & k_{FR} + k_{CA} + k_{EM} & -k_{CA} & 0 & 0 & k_{FR} \\ 0 & -k_{CA} & k_{CA} + k_{PA} & -k_{PA} & 0 & 0 \\ 0 & 0 & -k_{PA} & k_{PA} & 0 & 0 \\ -k_{CW} & 0 & 0 & 0 & k_{CW} & -k_{CW} \\ k_{CW} - k_{FR} & k_{FR} & 0 & 0 & -k_{CW} & k_{MA} + k_{FR} + k_{CW} \end{bmatrix} \quad (11)$$

where:

$$\frac{1}{k_{FR}} = \frac{1}{n^2} \left(\frac{1}{n_R \frac{EA}{l_f(t)}} + \sum_{i=1}^{n_{df}} \frac{1}{k_{di}} + \frac{1}{k_{fe}} \right); \frac{1}{c_{FR}} = \frac{1}{n^2} \left(\frac{1}{2\zeta \sqrt{\frac{n_R EA}{l_f(t)}} (m_{FR} + m_{CA} + m_{PA})} + \frac{1}{c_{fe}} \right) \quad (12a)$$

$$\frac{1}{k_{CW}} = \frac{1}{n^2} \left(\frac{1}{n_R \frac{EA}{l_c(t)}} + \sum_{i=1}^{n_{dc}} \frac{1}{k_{di}} + \frac{1}{k_{ce}} \right); \frac{1}{c_{CW}} = \frac{1}{n^2} \left(\frac{1}{2\zeta \sqrt{\frac{n_R EA}{l_c(t)}} m_{CW}} + \frac{1}{c_{ce}} \right) \quad (12b)$$

are generalized from previous expressions (2-3) due to the facts that in a $n:l$ roping installation the rope extension of the frame sling Δl_f , for example, is equal to n times the frame displacement y_{FR} , the suspended car weight P is n times the force exerted by the rope sling N ($P=nN$) and that the equivalent stiffness is the car weight divided by its displacement ($k_{FR}=P/y_{FR}$). Then the “in series” effect of the end terminals and the diverter pulleys, present in most elevator installations, are added. The terms k_{fe} and c_{fe} are the stiffness and damping of the frame sling end terminals, respectively; k_{ce} and c_{ce} are the stiffness and damping of the counterweight sling end terminals, respectively; n_{df} and n_{dc} are the number of diverter pulleys in the frame sling and counterweight sling, respectively; and k_{di} is the stiffness of the i -th pulley which deviates the rope sling.

The effect in the stiffness of the i -th diverter pulley is shown in figure 4. The rope is wrapped around the pulley with the angle of wrap β_i . The isolated pulley is tensioned by the rope traction P/n . If the pulley is fixed at the middle of a hinged-hinged axle beam of length l_{di} , 2nd moment of area I_{di} and modulus of elasticity E_{di} , the deflection of the beam/axle at the mid span f is given by:

$$f = 2 \cdot \frac{P}{n_R} \cdot \sin\left(\frac{\beta_i}{2}\right) \cdot \frac{l_{di}^3}{48E_{di}I_{di}} \Rightarrow \frac{1}{k_{di}} = \frac{\Delta l}{P/n} = \frac{2f \sin(\beta_i/2)}{P/n} = \frac{4 \sin^2\left(\frac{\beta_i}{2}\right) l_{di}^3}{48E_{di}I_{di}} \quad (13)$$

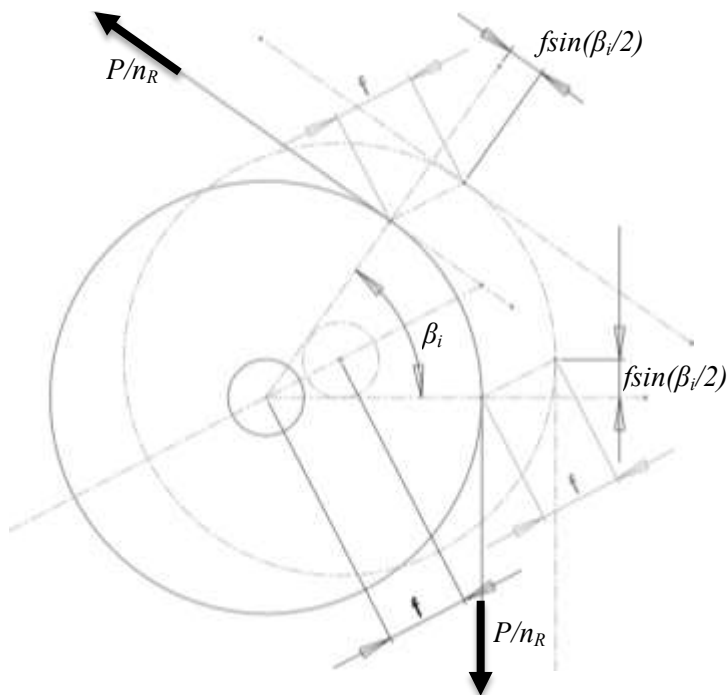


Figure 4. Derivation of the wrap angle β_i diverter pulley stiffness from the deflection f .

During normal operation both spring/damper for emergency actuation k_{EM}, c_{EM} are kept to zero (no actuation of the buffer nor the safety gear nor car-frame impact against a guide due to guide misalignment occur).

Finally, the applied forces vector $[F]$ under the presence of gravity of intensity g has the form:

$$[F] = \begin{bmatrix} \frac{T}{R} + \frac{(m_{mc} - m_{mf})g}{2} \\ -(m_{FR} + m_{mf}/2)g \\ -m_{CA}g \\ -m_{PA}g \\ (m_{CW} + m_{mc}/2)g \\ (m_{MA} + m_{mf}/2 + m_{mc}/2)g \end{bmatrix} \quad (14)$$

where T is the applied torque by the drive on the traction sheave which includes the motor torque and the torque applied by the brakes that makes it the rotor to stop rotation. The force exerted by the frame sling mass m_{mf} and counterweight sling mass m_{mc} on the frame and the counterweight, respectively, include the factor $1/2$ for equal static components deformation of the real system and the modeled discrete system in the reference configuration which is coincident with that of the system just after the putting on tension process. Then, the remaining mass of the slings $(m_{mf} + m_{mc})/2$ fixed to the machine (MA) has to contribute to the vertical displacement and gravity torque of the machine. The total static torque exerted on the traction sheave is the sum of the momentum applied by the drive torque T and the gravity torque: $(m_{mc} - m_{mf})gR/2$.

5. Results and Conclusions

Now we are ready to compare the performance of the various roping systems at the functional differences in the elevator installation. This is based on the analysis of various parameters in expressions (7-14) and has not involved solving the equations (7) which is undertaken in [14].

Table 1. Comparison of mechanical parameters for a number of roping systems. N_1 : permissible tension of one rope. T_0 : minimum drive torque required. k_{1fe} : spring constant of 1 rope end. k_{d1} : spring constant of 1 diverter pulley.

Roping	1:1	2:1	4:1	n:1
$(m_{FR} + m_{CA} + m_{PA} + m_{mf})g/N_1$	n_R	$2n_R$	$4n_R$	$n \cdot n_R$
$(m_{FR} + m_{CA} + m_{PA} + m_{mf} - m_{CW} - m_{mc})gR/T_0$	1	2	4	n
$\alpha R / \gamma_{FR}$ (permanent regime)	1	2	4	n
(Necessary rope length)/h	n_R	$2n_R$	$4n_R$	$n \cdot n_R$
$[k_{FR}(\text{rope end factor})]/k_{1fe}$	n_R	$2^2 n_R$	$4^2 n_R$	$n^2 \cdot n_R$
$[k_{FR}(\text{rope factor top drive})]/(EA/h)$	n_R	$2n_R$	$4n_R$	$n \cdot n_R$
$[k_{FR}(\text{rope factor bottom drive})]/(EA/h)$	$n_R/2$	$2^2 n_R/3$	$16n_R/5$	$n^2 n_R/(n+1)$
Min number of div pulleys ($\beta_i = \pi$)	0	2	6	$2(n-1)$
$[k_{FR}(\beta_i = \pi \text{ div pulleys factor})]/k_{d1}$	-	4	16/3	n^2/n_{df}
Min number of div pulleys ($\beta_i = \pi/2$)	0	4	12	$4(n-1)$
$[k_{FR}(\beta_i = \pi/2 \text{ div pulleys factor})]/k_{d1}$	-	1	16/6	n^2/n_{df}
Min $[M](1,1)(\text{div pulleys factor})/(I_d/r_d^2)$	-	2	6	$2(n-1)$

From table 1 above it is obvious that increasing roping ratio n reduces the minimum torque of the machine drive and the permissible load of the required rope but the necessary rope length is increased to the same amount. The permissible load of the required rope can also be reduced by increasing the

number of ropes n_R . The roping ratio n influences, mainly, the dynamics of the installation. The angular speed of the motor drive is increased by n and the startup and shutdown of the elevator will relax by the inherent increase in the number of diverter pulleys. However, the passengers comfort which is influenced by the stiffness of the suspension ropes remains almost constant for equal permissible tension of the rope design.

The formulas for generalized roping of the elevator has been derived. The proposed 6 d.o.f. system has been able to accommodate the performance of complex $n:1$ roping elevator systems. The simplicity of the proposed model is adequate to sensitivity analysis and assist the design process in lift engineering and education applications. The proposed model is implemented in ad-hoc software called ElevaCAD that implements the Kutta-Medem algorithm to get the solution of the governing equations by numeric time iteration to any excitation of the system and speed control strategy [14].

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